# Research on vibration reduction test and frame modal analysis of rice transplanter based on vibration evaluation

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**Abstract:** The chassis of rice transplanter tends to vibrate severely in the severe working environment, causing a severe effect on the operational performance and driving comfort. In order to avoid this situation, this paper constructs a vibration evaluation system of the rice transplanter and carries out experimental analysis. According to the optimal acceleration sensor placement scheme, a test platform system was designed. Taking the high-speed transplanter chassis as the research object, this study carried out the experiments modal analysis and optimization on the chassis. The three-dimensional model of the transplanting machine chassis established by SolidWorks was imported into ANSYS Workbench for finite element modal simulation analysis. Comparing the two modal analyses, it is found that the results data of the two analysis methods were very close. After optimization, the length $x_1$, the section width $x_2$ and the thickness of the hollow beam $x_3$ of the main load-bearing beam of the frame were as follows: $x_1$=1641.5 mm, $x_2$ =26.7 mm, $x_3$=5 mm, respectively. The maximum overshoot of the low-level system was reduced by 28.57%. It has been verified that the vibration of the whole machine has been effectively reduced.

**Keywords:** chassis, high-speed rice transplanter, vibration evaluation, modal analysis
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## 1 Introduction

The chassis, as the carrier of vehicle, is an important component of vehicle. The dynamic and static performance of the chassis have a direct effect on the vehicle working stability and life time[1,2]. Likewise, the walking chassis is also an important component of the high-speed transplanter. The high-speed transplanter may be caught in pit, slipping or have to pass a ridge when working in paddy fields[3,4]. Under the action of road load and continuous power output of the engine, the transplanter chassis will be subjected to strong impact, causing vibration[5]. Resonance will occur when the inherent vibration frequency of the chassis is close to the external load frequency. The resonance has a strong impact on the operating performance and driving comfort of the transplanter, and may even cause fatigue fracture, ultimately leading to an accident. Therefore, it is of great significance to effectively evaluate the vibration conditions of the whole machine and analyze the vibration characteristics of the transplanter chassis. Thus it's meaningful to optimize the design of its structure to make it equipped with necessary conditions for proper functioning.

Researches in China and other countries mainly focus on the transplanting mechanism, seedling pushing planting mechanism, transmission, etc. of transplanter[6,7]. Few studies put their emphasis on the vibration characteristics of the transplanter chassis. The countries that have thorough research on the main components of transplanters include Japan, South Korea, and the United States. The flat terrain and large land areas in these countries grant transplanters with high adaptability[8,9]. However, the widespread hilly areas and small and scattered lands in China provide a harsh working environment for transplanters, and the external loads suffered during the work and their own vibration are relatively strong. The external load is transmitted to each working component exactly through the chassis. Therefore, the whole machine vibration evaluation and vibration analysis of the transplanter's chassis are required.

The modal analysis methods of vibrational structure include test modal analysis and finite element modal analysis[10,11]. Modal test is often used to verify the correctness of finite element analysis. Considering the economy and accuracy of test analysis, how to determine the optimization object and design plan has always been an important issue.

A method for evaluating the vibration of the whole rice transplanter is proposed in this study. The vibration reduction effect and the accuracy of the evaluation results were verified by experimental tests. A finite element model was built for the 2ZG-6DK high-speed transplanter chassis, and then the modal analysis was carried out[12,13]. The three-dimensional modeling software was used to build a model of the chassis. And the ANSYS software was used for stress analysis. And then the modal inherent frequency and vibration mode of the transplanter

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chassis were also obtained. The obtained data were compared with the modal test results to demonstrate the effectiveness and accuracy of the finite element modal analysis method. Based on the damping characteristics of the transplanter chassis, the dynamic output response curve of the chassis was calculated. It was compared with the optimized dynamic response, proving the correctness of optimizing the structure of the rice transplanter chassis.

## 2 Vibration evaluation of the whole rice transplanter

The mechanical vibration generated during the operation of the rice transplanter will affect the comfort and health of the driver. The shock vibration transmitted through the seat was the main source of vibration of agricultural machinery. In order to improve the working environment, it is of practical significance to take appropriate measures to reduce mechanical vibration. Prior to this, appropriate methods should be adopted to make a reasonable evaluation of the vibration conditions of the agricultural machinery. It is used to estimate the risk of agricultural machinery operation and operation.

### 2.1 Theoretical basis of vibration evaluation

The vibration situation of rice transplanter was complicated, including periodic, random or transient mechanical vibration. The common engineering standard ISO2631-1:1997 evaluates vibration. GB/T13441.1-2007 was equivalent to adopting this international standard. According to the "Mechanical vibration and shock - Evaluation of human exposure to whole-body vibration - Part 1: General requirements" a set of vibration evaluation systems for the whole rice transplanter was designed using the weighted root mean square acceleration method[14-17].

Acceleration is the most commonly used measurement index for mechanical vibration, which is of great significance for evaluating the vibration status of the whole machine. In order to reasonably and accurately evaluate the vibration condition of the whole rice transplanter, it is necessary to detect and analyze the acceleration of multiple measuring points of the rice transplanter. Since the acceleration changes in the three rotation directions of the rice transplanter were small during operation, only the influence of the three translational accelerations needs to be considered. The three-axis acceleration sensor was used to obtain acceleration data for each measuring point. And the weighted acceleration was $a_{wi}$.

$$a_{wi} = \left[\sum_i \left(w_i a_i\right)\right]^{\frac{1}{2}} \tag{1}$$

where, $a_{wi}$ is the frequency weighted acceleration, m/s$^2$; $w_i$ is the $i^{th}$ weighting factor; $a_i$ is the $i^{th}$ root mean square acceleration, m/s$^2$. The weighted root mean square acceleration of each axis in the time domain was calculated according to the following equation:

$$a_w = \left[\frac{1}{T}\int_0^T a_{wi}^2\left(t\right)dt\right]^{\frac{1}{2}} \tag{2}$$

where, $a_w$ is the weighted acceleration in the time domain, m/s$^2$; $T$ is the test duration, s.

When solving the vibration evaluation index value, it is necessary to synthesize the vibration data in different directions, that is, to solve the total amount of vibration. In the orthogonal coordinate system, the root mean square value of the weighted acceleration was solved and the total amount of vibration was synthesized to obtain the final vibration evaluation index $a_v$.

$$a_v = (k^2{}_x a^2{}_{wx} + k^2{}_y a^2{}_{wy} + k^2{}_z a^2{}_{wz})^{\frac{1}{2}} \tag{3}$$

where, $a_v$ is the total amount of vibration synthesized by the vibration components in each direction, m/s$^2$; $a_{wx}$, $a_{wy}$, and $a_{wz}$ are the root mean square value of weighted acceleration on $x$, $y$, and $z$ orthogonal coordinate axes respectively, m/s$^2$; $kx$, $ky$, and $kz$ are the direction factors on the corresponding coordinate axis respectively, and $k_x$=$k_y$=1.4, $k_z$=1.0 since the rice transplanter operator is in a sitting position when working.

**Table 1 Acceleration index value and vibration evaluation**

| $a_v$/m s$^{-2}$ | Vibration evaluation | $a_v$/m s$^{-2}$ | Vibration evaluation |
|---|---|---|---|
| <0.315 | Excellent | 0.8-1.6 | Pretty bad |
| 0.315-0.63 | Good | 1.25-2.50 | Very bad |
| 0.5-1.0 | Poor | >2.0 | Extremely bad |

After the acceleration was calculated by weighting according to the frequency range, the maximum value of the calculated acceleration root mean square value was compared with the vibration evaluation threshold to evaluate the vibration condition. Table 1 was determined according to the current standard GB/T 13441.1-2007 "Mechanical vibration and shock–Evaluation of human exposure to whole-body vibration–Part 1: General requirements" Section C2.3 "Comfort Response to Vibration Environment".

Finally, the calculated value of the vibration acceleration index of the rice transplanter was compared with the vibration evaluation threshold value in Table 1 to evaluate the vibration status. Thus, the vibration characteristics of the whole rice transplanter can be obtained, and the vibration comfort of the transplanter can be evaluated, that is, the vibration condition of the whole rice transplanter can be evaluated.

### 2.2 System composition

The vibration evaluation system of rice transplanter combines hardware test equipment with software analysis. It was mainly composed of signal acquisition module, data analysis and evaluation module, and so on. The signal acquisition module mainly includes a triaxial acceleration sensor of Model 356A16 with a measuring range of 50 g, a sensitivity of 100 mV/g. The dynamic signal acquisition instrument is Model DH5902N with the highest sampling frequency of 100 kHz, and the vibration magnitude within 20 g, which has the function of wire/wireless communication. Data acquisition system adopted the dynamic signal acquisition instrument of LMSSCADAS Mobile, and the software analysis was mainly carried out through the vibration analysis and evaluation module of the DHDAS software, which can monitor the vibration status of the whole rice transplanter in real-time. The test equipment that constitutes the vibration evaluation system of the rice transplanter is shown in Figure 1.


a. Accelerometer

b. Dynamic signal acquisition instrument

Figure 1 Modal test equipment

### 2.3 Experiments

#### 2.3.1 Principle of the experiments

First, multiple sensors were used to collect vibration data for each key working part of the rice transplanter. Then different weights were assigned to accelerations of different frequencies to find the root mean square value and the maximum value of the root mean square value was taken as the final index of vibration evaluation. The modal test procedure is shown in Figure 2.

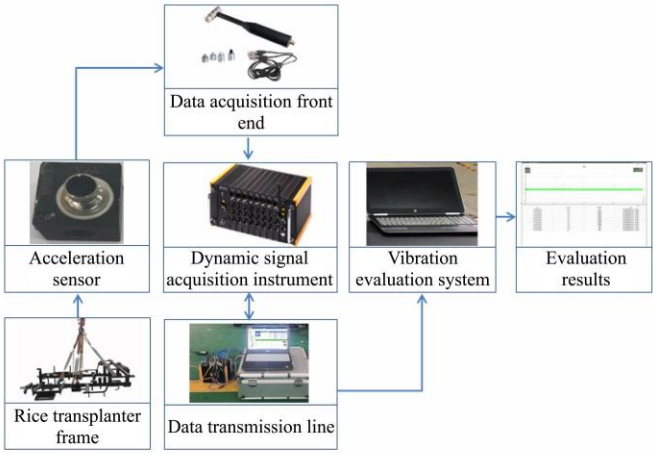



Figure 2 Modal test procedure

2.3.2 Sensor layout

The central coordinate axis was defined as follows: the front direction of the rice transplanter was the *y*-axis, the *x*-axis was on the horizontal plane and perpendicular to the *y*-axis, and the *z*-axis was the vertical direction. Sensors were arranged for multi-source vibration data collection and the sensor coordinate axis was aligned with the center coordinate axis accurately. In the experiment, each acceleration sensor was connected to 3 data acquisition transmission lines. Each line corresponds to 1 analog channel on the multi-function signal acquisition instrument. The 4 groups of sensors were arranged orthogonally with their bases respectively to fix them at the designated positions of the rice transplanter. The data acquisition front end was connected to the sensor and the dynamic signal acquisition instrument respectively.

2.3.3 Experiment method

First, check whether the parts of the rice transplanter are in good condition before the test. Evaluate the test conditions to ensure that the failure of the rice transplanter itself causes interference and errors in the test. Arrange three-way acceleration sensors at key positions of the rice transplanter and connect them to the multifunctional signal acquisition equipment in an orderly manner. Mark the location of each sensor and the name of the connected channel. After installing the sensor, data acquisition front-end line, notebook, and other test instruments, open the notebook to start the multi-source vibration evaluation system, set the sampling frequency to 1 kHz, the measurement type was acceleration measurement, and the sensor type was the charge output piezoelectric acceleration sensor.

After the rice transplanter equipment was started, in order to obtain a stable random vibration signal, after its running state was stable, click the “Acquisition” button to collect data. Click “Stop” after the collection is over. The length of the test time for each group of stable speed is 3min. After the vibration data at a certain speed was measured, the engine speed was changed and the signal was collected again in this way. The engine speed of the rice transplanter was from low to high. Five groups of speeds for the vibration test were set. Vibration tests were carried out at four positions (engine, frame, planting arm and rice transplanter seat) at 5 sets of speeds. Record the data separately. In the actual test, the actual engine speed obtained by the non-contact laser tachometer were 1547 r/min, 2072 r/min, 2405 r/min, 2885 r/min, and 3496 r/min.

After the test, the data was imported into the vibration evaluation module of the rice transplanter for processing. The data was extracted on site and post-processed through DHDAS system to achieve the purpose of vibration evaluation.

### 2.4 Vibration evaluation results and analysis

A certain high-speed rice transplanter was taken as the research object in this experiment. Acceleration sensors were arranged at the positions of its engine, seedling planting arm, seedling claws and chassis frame respectively, and vibration evaluation software was used to evaluate its vibration.

a. Planting arm b. Engine connection c. Seedling carrier d. Frame

Figure 3 Measuring point of vibration test

It can be seen from the vibration data of the rice transplanter that there were many vibration data of the rice transplanter and its curve had several large protrusions. It showed the poor vibration of the rice transplanter.

Figure 4 is the data collection diagram of the vibration test of the whole machine. The results showed that a large amount of vibration data will be generated during the operation of the rice transplanter. And it can be seen that some locations where the vibration data was abnormally increased, which provided a factual basis for determining the position of the rice transplanter with poor vibration.

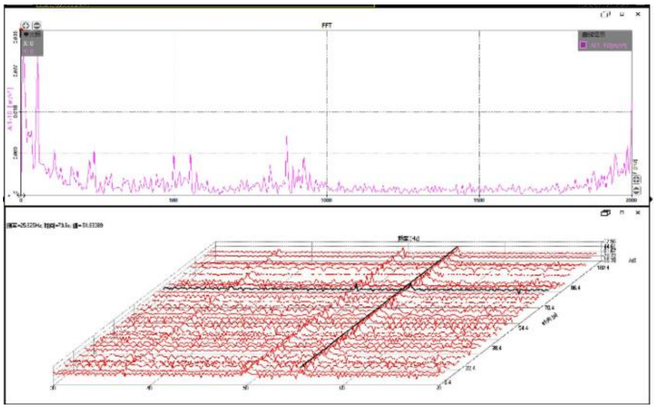


Figure 4 Vibration data acquisition

Through the test and evaluation of the vibration condition of the whole rice transplanter, it was found that the vibration condition of the whole machine was “pretty bad/very bad”. And the place with large vibration was channel AI1-03 (sensor 4), that is, the position of the rice transplanter frame. This showed that the frame of the rice transplanter become the main source of vibration under this working condition. Therefore, it is of practical value to carry out research on the vibration characteristics of the frame of the high-speed rice transplanter and to carry out research on the vibration reduction of the whole machine on this basis.

## 3 Vibration characteristic analysis

### 3.1 Experimental subject

The structure of the ride-on style high-speed transplanter chassis is shown in Figure 6. When working in paddy fields, the transplanter chassis often is subjected to unstable loads and cushioning forces, which directly leads to strong vibration of the chassis. Fatigue fracture to the chassis caused by vibration is an uncontrollable factor that destroys the chassis[19]. Therefore, a monitoring means and an analysis method are required to monitor and evaluate the rice transplanter chassis, so as to optimize the structure and provide a reference for the design of the suspension vibration absorption system.

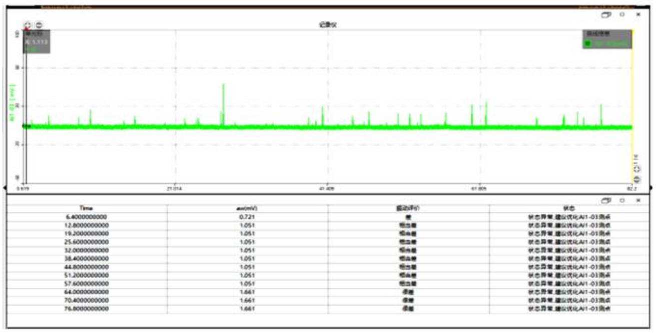


Figure 5 Vibration evaluation results

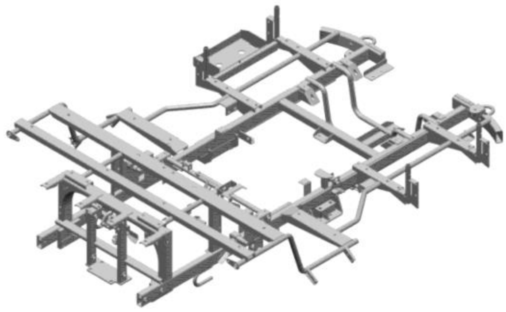


Figure 6 High-speed rice transplanter chassis

The changeable working environment also has different effects on the load distribution, deformation, stress, rigidity and other characteristics of the rice transplanter chassis. In addition, driving safety and comfort are also essential factors to be considered during operation of the transplanter. Thus, vibration analysis is required for the transplanter chassis.

**3.2 Building of finite element model of chassis**

The high-speed transplanter has a chassis made of spatial thin-walled beams structure. The chassis was composed of a non-loaded spatial boundary beam with variable heights, two main longitudinal beams with hollow rectangular cross-sections, and several load-bearing transverse beams with annular cross sections. In view of such structural characteristics and the limitation that the beam element cannot reflect the stress distribution in the joint area of the transverse beam and the longitudinal beam, the plate element is selected as the basic discrete unit in the modeling process. Then the idea of modularization is used to disperse the 3D model of the chassis. Based on this, a finite element model with a high degree of agreement with the chassis model was built[18-20]. Figure 7 shows the finite element discrete model of high-speed transplanter chassis.

**3.3 Finite element modal analysis of the chassis**

Mode is one of the important characteristics of mechanical structure vibration, so the modal analysis of the transplanter chassis is an effective way to study its vibration. The modal parameters of structural vibration include vibration mode, inherent frequency, matrix, etc., which can be obtained by modal analysis of the chassis. The material and attribute were entered the ANSYS software, and the inherent frequencies, vibration modes and cloud charts of vibration modes of the first ten orders of the chassis through simulation, as shown in Figure 8.

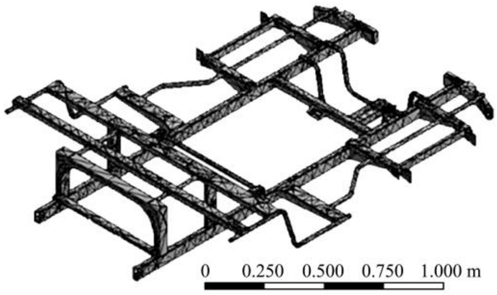



Figure 7 Finite element model of chassis

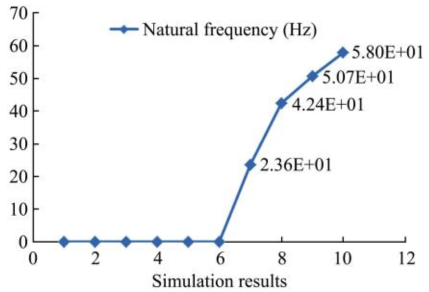



Figure 8 Simulation modal analysis results

According to literature research[21,22], the vibration frequency of the frame of the high-speed rice transplanter was close to the excitation frequency of the field ground unevenness and the vibration frequency of the main drive shaft, which is likely to cause resonance. This verifies the accuracy of the vibration evaluation system.

**3.4 Modal test**

In order to obtain the more accurate dynamic characteristics of vibration of the transplanter chassis, a modal test was required for verification based on the finite element modal analysis of the chassis. Modal optimization test was a dynamic test based on structure input and response output of the system. And it is often used to describe the dynamic characteristics of a structure. Verification of the accuracy of finite element analysis through test results was a key step in the analysis of structural vibration characteristics[23].

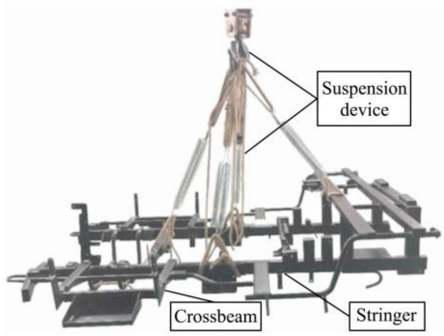



Figure 9 High-speed rice transplanter chassis test device

3.4.1 Equipment and test methods

The modal test for the vibration characteristics of 2ZG-6DK transplanter chassis was conducted on the DH5902 signal testing system, as shown in Figure 10.

During the test, the test points of the main load-bearing beam of the chassis, the suspension beam of the transplanting mechanism, the side support plate and the main support plate were divided according to their actual load conditions and then accordingly marked. A three-point suspension device was used to suspend the chassis and the traxial acceleration sensors were arranged on the points to be tested. An impulse hammer was used to simulate the compound load applied to the chassis by hammering each test point.

Force hammer, model 086C03, range from 0 to 5000 N, sensitivity 2.28 mV/N; The curvature strains in the *x*, *y*, and *z* directions at each test point were collected by the DH5902 signal test system, and the system response function was thus obtained. The vibration characteristics of the transplanter chassis were obtained after analyzed the response function by the modal analysis software on the computer. The schematic diagram of the modal test is shown in Figure 11.

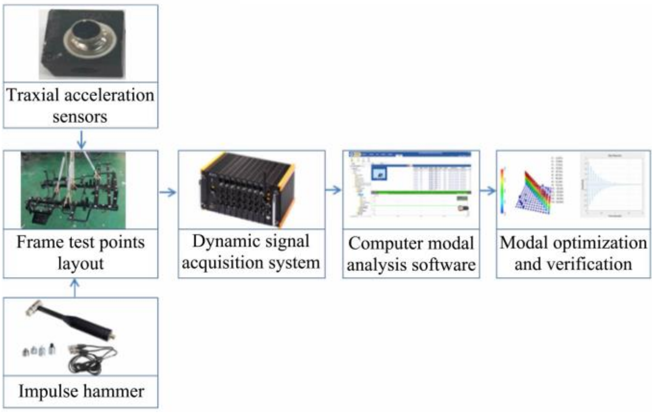



Figure 10 Schematic diagram of the modal test system

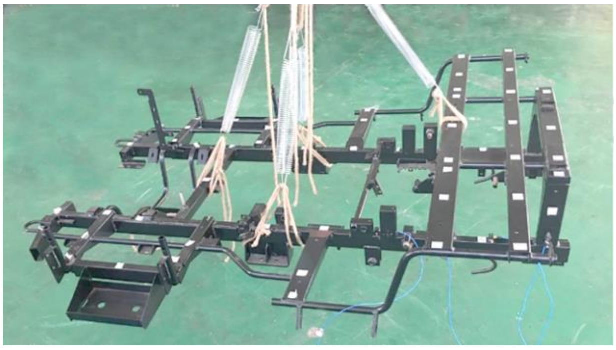


Figure 11 Test point layout

The sensors were respectively arranged on the key parts of the transverse beams, such as the junction between the beams, the middle position of the beam, and the welding point. Therefore, these test points were determined. The test points on the transplanter chassis were arranged as shown in Figure 11.

3.4.2 Modal test results

In order to completely reflect the vibration characteristics of the transplanter chassis and ensure accuracy, a free excitation method was used in this test to hang the chassis with springs by three points.

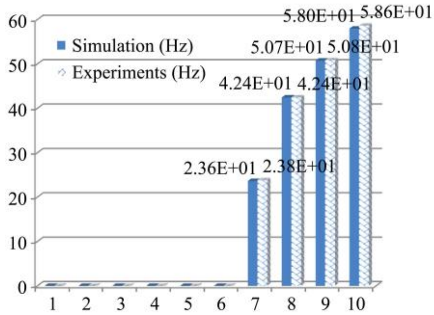



Figure 12 Comparison of simulation and test results

Modal test results showed that there was a small relative error between the measured frequency and the calculated value. The simulation modal parameters matched the experimental model. Since the low-order inherent frequency of the chassis has a great influence on its structural dynamic characteristics, based on this, it can be considered that the simulation results were highly correlated with the test results.

## 4 Optimization of frame structure parameters

### 4.1 Output response of frame system

For any system, there are many kinds of transient response indicators. The maximum overshoot $\sigma_p$% is an important indicator to evaluate the oscillation performance of a system. The stronger the oscillation of the system, the easier it is to cause structural fatigue damage due to resonance. If the vibration of the frame can be reduced, the vibration can be reduced. Therefore, the main goal was to reduce the maximum overshoot of its low-level system output response.

For a linear system with *m* degrees of freedom, its kinematics equation is as follows.

$$M\ddot{x}(t)+D\dot{x}(t)+Kx(t)=f(t) \tag{4}$$

where, *M*, *D*, and *K* represent the mass matrix, damping matrix and stiffness matrix, respectively. The transfer function based on the system modal parameters obtained after Laplace transform is as follows.

$$\frac{X_0(s)}{X_i(s)}=\frac{1}{Ms^2+Ds+K}=\frac{\omega_n^{\ 2}}{s^2+2\xi\omega_n s+\omega_n^{\ 2}} \tag{5}$$

in which, $\omega_n=\sqrt{\frac{K}{M}}$ is the natural oscillation frequency and $\xi=\frac{K}{2\sqrt{MD}}$ is the damping ratio. After the inverse Laplace transform of the transfer function of the system, the output time response of the system was:

$$X_0(t)=1-\left(e^{-\xi\omega_n t}\cos\omega_d t+\frac{\xi}{\sqrt{1-\xi^2}}e^{-\xi\omega_n t}\sin\omega_d t\right) \tag{6}$$

where, $\omega_d$ is the damped oscillation frequency. By analyzing the transient performance index of the time response of the system vibration output, it can be known that when the undamped oscillation frequency $\omega_n$ remains unchanged, the damping ratio $\xi$ increases, and the maximum overshoot of the system is smaller. That is, the system's oscillation performance was weak, and the relative stability of the system is good. The response time of the system will also become longer. That is, the sensitivity of the system is poor. On the contrary, if $\xi$ is reduced, the greater the maximum overshoot of the system, the more severe the oscillation of the system. That is, the system was more prone to structural damage at this frequency. Therefore, the maximum overshoot $\sigma_p$% of the output response of the system was analyzed, namely:

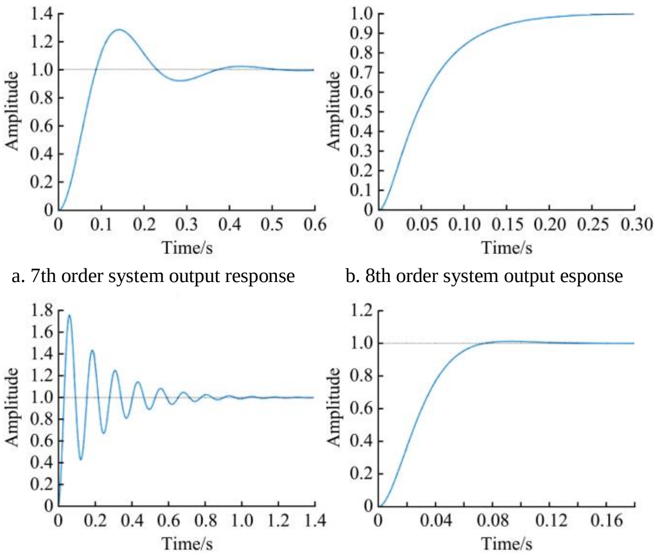
$$\sigma_p=\exp\left(\frac{-\xi\pi}{\sqrt{1-\xi^2}}\right)\times100\% \quad (0<\xi<1) \tag{7}$$

After further calculation, the result shown in Figure 13 was obtained.

Combining Table 2 and Figure 13, it can be seen that the 7th and 9th order system output response maximum overshoot $\sigma_p$% were both greater than 28.57%. The 9th order was more oscillating. It was more prone to structural damage caused by resonance.

**Table 2 The maximum overshoot of the system**

| Order | Mode 7 | Mode 8 | Mode 9 | Mode 10 |
|---|---|---|---|---|
| $\xi$ | 0.373 | 1.258 | 0.088 | 0.816 |
| $\sigma_p$ | 28.57% | 0 | 75.76% | 1.19% |



a. 7th order system output response b. 8th order system output esponse

c. 9th order system output response d. 10th order system output response

Figure 13 Output response of the frame before optimization

### 4.2 Optimal design

The length $x_1$, section width $x_2$ and thickness $x_3$ of the main load-bearing beam of the frame were optimized. Empirically select the damping ratio $\xi$ as 0.6-0.8. Based on the empirical value of $\xi$ and the deformation status of the modal test frame, the mathematical model of the rice transplanter frame was established by using the constraint optimization method:

$$\max(\xi) = \xi(X) = \frac{K(X)}{2\sqrt{M(X)D(X)}} \tag{8}$$

$$X = [x_1, x_2, x_3] \tag{9}$$

$$\text{s.t.} = \begin{Bmatrix} 1200 < x_1 < 1800 \\ 10 < x_2 < 80 \\ 2 < x_3 < 10 \end{Bmatrix} \tag{10}$$

After calculating through Matlab, we obtained $x_1$=1641.5 mm, $x_2$=26.7 mm, $x_3$=5 mm.

## 5 Experiment and discussion

### 5.1 Experiments

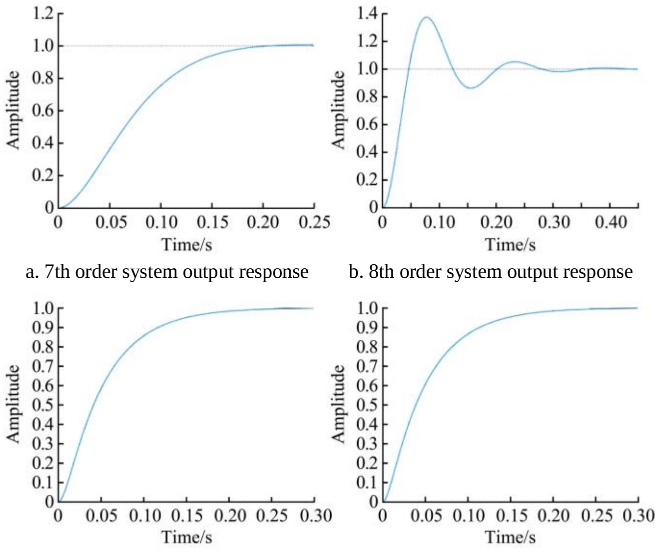
In order to optimize the vibration condition of the whole high-speed rice transplanter, according to related research[24-28], the work was done on the 2ZG-6DK rice transplanter support arm (planting arm) vibration reduction and engine vibration isolation. That is, the vibration of the rice transplanter was optimized through optimizing the structure parameters of the rice transplanter support arm and the engine double-layer vibration isolation system. On this basis, the frame was manufactured and installed and debugged according to the optimized frame structure parameters.

Combined with the optimized frame, support arm and engine vibration isolation system, the vibration evaluation system was used to evaluate the vibration of the rice transplanter again. The test plan was the same as the test setup in Chapter 2. By carrying out verification tests, comprehensive analysis and evaluation of the vibration conditions of the high-speed rice transplanter machine were carried out. And its optimization effect was verified.

### 5.2 Results

Bring $x_1$=1641.5, $x_2$=26.7, $x_3$=5 into $\xi$, and calculate the maximum overshoot again. The response curve shown in Figure 14 was obtained.

By optimizing the output response curve of the rice transplanter frame system and Table 3, it can be seen that the oscillation of the rice transplanter frame system under the 7, 9 and 10 orders was obviously weakened, especially the 7th order oscillation was basically eliminated. The rationality of the calculation and the feasibility of the damping optimization method were verified.



a. 7th order system output response b. 8th order system output response

c. 9th order system output response d. 10th order system output response

Figure 14 The output response of the frame after optimization

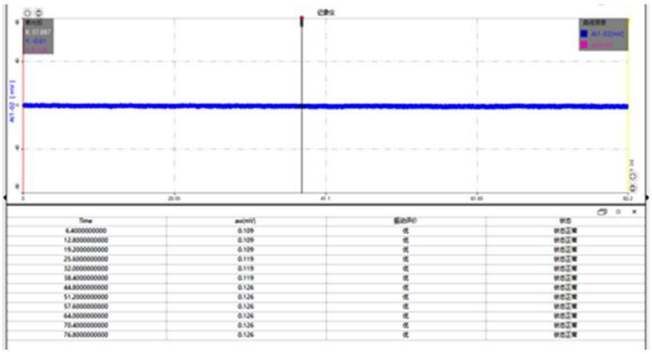


Figure 15 Vibration evaluation after optimization

**Table 3 The maximum overshoot of the system before and after optimization**

| Order | Mode 7 | Mode 8 | Mode 9 | Mode 10 |
|---|---|---|---|---|
| Before $\sigma_p$ | 28.57% | 0 | 75.76% | 1.19% |
| After $\sigma_p$ | 0 | 38.65% | 0 | 0 |
| Amount of change Δ | –28.57% | 38.65% | –75.76% | –1.19% |

### 5.3 Discussions

The theoretical modal analysis method in this paper had uncertainty when setting the physical parameters of the material. The determination and simulation of the boundary conditions of the model were also not realistic enough. The influence of structural damping was often ignored in theoretical modal analysis. The above factors will affect the study of the dynamic characteristics of the mechanical structure. The next research should improve the physical parameters of the material. At the same time, it should be discussed how to determine the model boundary conditions and simulate them to improve the accuracy of the analysis results.

Since only a set of optimized structural parameters of the rice transplanter frame were tested in the experiment, a variety of optimization schemes could not be adopted for comparative analysis. The next step is to determine some optimization schemes and develop entities for multi-faceted comparison. Field tests should be carried out to verify the rationality and accuracy of the frame structure parameters optimization design and related work.

## 6 Conclusions

(1) The vibration evaluation system of the rice transplanter was constructed and experimental analysis was carried out in this paper. It was found that the vibration of the whole rice transplanter was poor and the most violent position was the frame. It laid the foundation for the research on the vibration characteristics of the rice transplanter frame and the vibration reduction of the whole machine.

(2) The optimized design of the frame structure parameters avoided resonance. After optimization, the length $x_1$, the section width $x_2$ and the thickness of the hollow beam $x_3$ of the main load-bearing beam of the frame were as follows: $x_1$=1641.5 mm, $x_2$=26.7 mm, $x_3$=5 mm, respectively.

(3) Vibration on the transplanter frame can be reduced. The vibration evaluation system of the rice transplanter was used to test and verify the vibration of the whole machine. The maximum overshoot of the low-level system was reduced by 28.57%. It was found that the optimized vibration evaluation result was all "excellent", which achieved the purpose of vibration reduction of the whole machine.

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